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Ejector Configuration for Designing a Simple and High Performance Solar Cooling System

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Abstract

Ejector refrigeration cycle is a simple system, which can provide cooling by using solar thermal energy. Instead of a mechanical compressor of refrigeration cycle, refrigerant in evaporator is evacuated by an ejector using supersonic flow generated by the vapor pressure at temperatures being higher than around 60 °C. The ejector configuration design is a key to get a high efficiency of the cycle. In order to realize the behavior of the refrigerant in ejector, computational fluid dynamics (CFD) is applied to get the best geometry parameters, and then the parameters are confirmed by experimental trials. The CFD is developed as an in-house solver of the compressible Navier-Stokes equations. Through the numerical and experimental approach, four sets of configuration parameters of a mixing section area and a nozzle exit area of the ejector are being considered. A conclusion is that an appropriate mixing section area can make the greater cooling capacity but the condensing temperature decreases, while a smaller nozzle exit area can avoid the energy loss of shockwave that makes reliable repeatability and higher condensing temperature but lower cooling capacity in our experimental trials. The performances of ejector cycles having four different sets of configuration parameters cannot exceed a linear relationship between cooling capacity and condenser temperature so far. This paper reports an up-to-date result for developing an appropriate design method of ejector configuration.

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Keywords: ejector; refrigeration cycle; CFD; mixing section; nozzle exit

1. Introduction

Development of cooling system working with thermal energy can avoid a rapid increase of electricity demand. We have studied an ejector refrigeration cycle, which could provide cooling by using solar thermal energy. Because an ejector itself is a key component in this cycle to get higher performance by

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the suction ability, the ejector configuration design is important. Analysis of the behavior of refrigerant flow in the ejector is an effective way to get an appropriate ejector configuration. The computational fluid dynamics, CFD, using an in-house solver of the compressible Navier-Stokes equations and the experimental tests for confirming an actual cycle performance are being conducted.

Nomenclature

P	pressure [Pa]	Subscript	
T	temperature [K]	g	generator
Q	heat rate [W]	c	condenser
W	power, electricity [W]	e	evaporator
m	mass flow rate [kg·s^{-1}]	i	inlet
h	enthalpy [J·kg^{-1}]	o	outlet
COP	coefficient of performance [-]	p	pump
η	efficiency [-]	NE	nozzle exit
A	area ratio to throat area ratio [-]	MIX	mixing section
l	length [mm]	Superscript	
		*	critical point

2. Ejector refrigeration cycle

From the description by Nakagawa et al. [1], the idea of ejector refrigeration cycle was proposed in the early 1900s and a steam-jet refrigerator was firstly developed by Leblanc in 1910 as the actual system. Many academic papers concerning the ejector cycle have been still reported. An advantage would be that an ejector refrigeration cycle is simpler system than absorption or adsorption systems, while the thermal efficiency is lower than those. Figure 1 shows a schematic principle of a single ejector refrigeration cycle, which is used as an experimental test device in this study. This cycle can work by heating a liquid refrigerant in vapor generator. The higher pressure and higher temperature refrigerant vapor produces a primary or a driven flow in an ejector, which can suction refrigerant from an evaporator. These primary and suction flows merge and pass through a mixing section and a diffuser as shown in Figure 2, and the mixed flow gets into a condenser to be condensed. A part of liquid refrigerant is sent back to the vapor generator by a liquid pump. Other part of liquid in the condenser comes to an expansion valve and evaporates in an evaporator. Three different thermal energy levels are required in an ejector cooling cycle, i.e., generator, condenser, and evaporator, which is different from a vapor-compression heat pump system having two thermal energy levels of condenser and evaporator. When condenser temperature becomes an ambient temperature in ejector refrigeration cycle, the thermal energy required becomes two levels of vapor generator and evaporator.

3. Numerical analysis for optimum flow

Figure 2(L) shows a schematic of an ejector. An ejector consists of four parts, nozzle section, suction chamber, mixing section, and diffuser. The vapor flow after the throat area in ejector becomes supersonic at a certain condition of a nozzle inlet pressure, a generating pressure P_g, and a backpressure of the diffuser, i.e., condensing pressure P_c. Figure 2(R) shows parameters of an ejector in the calculation. The parameters in calculations are a condensing pressure P_c, a mixing section area ratio A_{MIX}, and a nozzle exit area ratio A_{NE}. Both area ratios are non-dimensionalized by the nozzle throat area. The diameters of the nozzle throat and a mixing section are important in the ejector to be designed. Chan et al. [2] [3], members of our group, reported the analytical predictions of these dimensions and the preliminary design of an ejector configuration based on the thermodynamic relations of shock-circle model [3].

In addition, a computer fluid dynamics, CFD, is applied for the designing in this study. We reported the calculation results of the parameters of an ejector [4]. An entrainment ratio ω as given in equation (1) was used as an evaluation indicator of cycle efficiency, where the m_{go} is the mass flow rate of a driven flow and the m_{eo} is the mass flow rate of suction flow.

$$\omega = \frac{m_{eo}}{m_{go}} \tag{1}$$

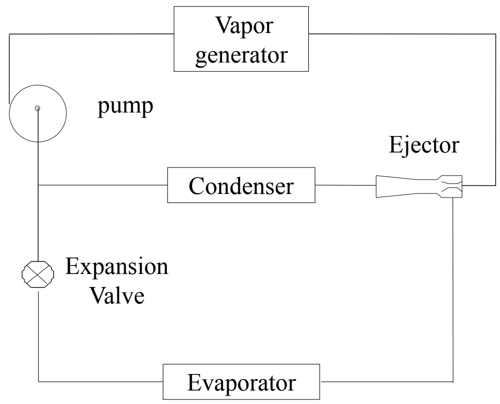

Fig. 1 A schematic of ejector refrigeration cycle

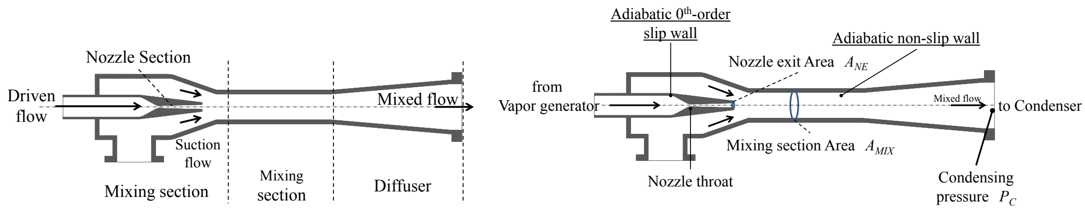

Fig. 2 (L) Schematic of an ejector; (R) Parameters of an ejector in calculation

4. Experimental set up

4.1. Experimental testing device

For a purpose of measuring the actual performance of an ejector cycle, an indoor testing device was set up. Figure 3 shows a schematic of the whole set of testing device, which consists of a vapor generator, a condenser, an evaporator, a liquid receiver, a liquid pump, and an ejector. Heat source of the vapor generator, which will be supplied by solar thermal energy in an actual case, is simulated with a circulating bath that can supply heat at controllable temperatures up to 200 °C with a precision of 0.01 °C. In similar manner, a temperature of the condenser is simulated by a cooling bath.

The refrigerant HFC-134a was selected from the thermodynamic properties among propane, isobutane, n-butane, and HFC-134a as explained in a previous paper [2]. Although propane is expected to have better performance than HFC-134a, HFC-134a was selected due to safety reason.

All the temperature measurements at the points as shown "T" in Figure 3 are K-type thermocouples. The flow rates of a heat transfer fluids, water, in circulating-baths for vapor generator and evaporator are measured by using Keyence FD-V70 series flow-sensors with sensor head FD-P20, whose range of detection is 2-20 L/min and the repetitive accuracy is better than 0.1 % of F.S. The pressure of working fluid is measured at "P" in Figure 3 by using Keyence AP-V80 series pressure-sensors with the sensor heads AP-14S and AP-13S. The range of detection is 0-10 MPa, and the repetitive accuracy is 0.1 % F.S.

An operating procedure of the device is simple. Firstly, the circulating thermostat baths for vapor generator and a cooling bath for condenser are switched on. When heat is provided to the generator, refrigerant boils and vapor is generated and flows at sound speed at the thread of nozzle in ejector. Then we can measure the performance. The condenser pressure is a function of the temperature. The liquid pump operation was controlled manually with an inverter speed controller by observing the liquid level in receiver tank through a glass. The vapor from a diffuser of the ejector is cooled in water coming from the cooling bath and changes from vapor phase to liquid phase. After heat-exchange at condenser, a part of water flows as the cooling load in evaporator and then water flows through the cooling bath and come back to condenser again.

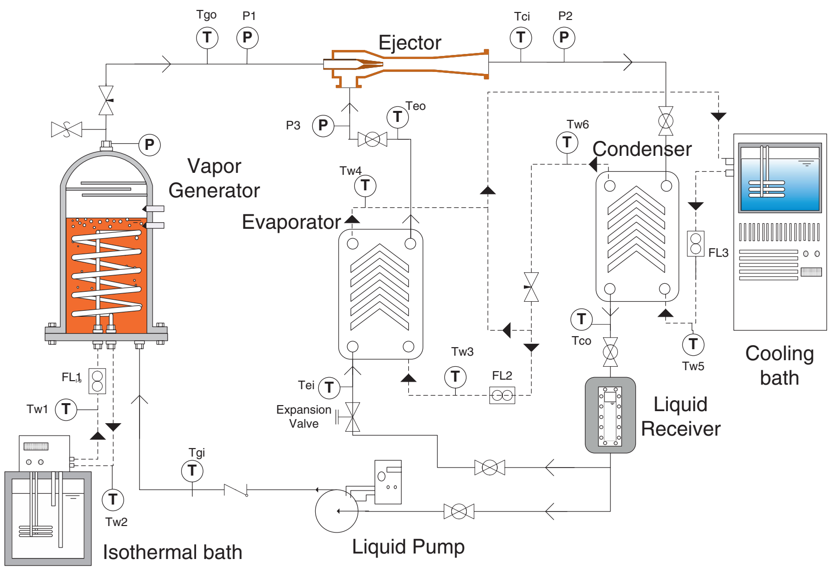

Fig. 3 A whole set of experimental testing device

4.2. Evaluation indicator in experiments

In order to evaluate the cycle performance in experiments, thermal COP, COP_T, was used as an evaluation indicator. Equation (2) shows relationship between COP_T and ω. That implies COP_T is proportional to entrainment ratio when both T_g and T_e are constant. The entrainment-ratio-increase results a linear increase of the efficiency. The COP_T is calculated from measuring thermal input to vapor generator Q_g, and cooling capacity in evaporator Q_e. The Q_g is obtained by measuring a water flow-rate and a temperature change between inlet and outlet of isothermal bath. The Q_e is also obtained as the same way at cooling bath. Performance of the cycle

$$\mathrm{COP_T} = \omega \times \frac{h_{eo} - h_{ei}}{h_{go} - h_{gi}} = \frac{\text{cooling capacity}\, Q_e}{\text{heat input}\, Q_g} \tag{2}$$

4.3. Effect of mixing-section diameter

The $\mathrm{COP_T}$ of an ejector-cycle is constant at a temperature of condenser being lower than the critical condensing temperature, T_c^*. At the condition, the driven- and suction-flows in ejector are supposed to be choked at sound speed, which means the entrainment ratio ω and $\mathrm{COP_T}$ are also a certain constant values. On the other hand, $\mathrm{COP_T}$ drastically decrease when the condensing temperature becomes higher than T_c^*.

Figure 4 and Table 1 show two sets of the configuration of ejectors A and B. The ejector A was designed by Chan et al. in our group, which has a mixing-section diameter of 2.36 mm and the performance was about 0.35 for ω with T_c^* of about 27°C at a condition of T_e = 15 °C and T_g = 60 °C as reported in papers [2] and [3]. The $\mathrm{COP_T}$ of ejector A is about 0.4, which is a linear function of ω as explained above. The ejector B, which has a mixing-section diameter of 3.00 mm, was designed based on the result of numerical analysis. The $\mathrm{COP_T}$ of ejector B was about 0.85, which reached as double as that of ejector A. On the other hand, as shown in Fig. 5, the comparison of experimental and numerical analysis for ejector B at the same condition of T_e = 15 °C and T_g = 60 °C, the ω of the experiment is slightly higher than the numerical analysis result, but the critical pressure is lower by approximately 0.1 MPa than that of the numerical analysis result.

In order to apply an ejector refrigeration cycle to a solar powered air-conditioning system, T_c^* should be close to an ambient temperature such as 30 °C for keeping a stable operation in summer. The critical pressure of condenser measured by using ejector B was about 0.6 MPa, which corresponds to about 21 °C of T_c^*. On the other hand, from numerical analysis it was calculated as about 0.7 MPa, which corresponds to about 27 °C of T_c^*. We are again requested to improve the critical pressure to be higher about 0.1 MPa for ejector B.

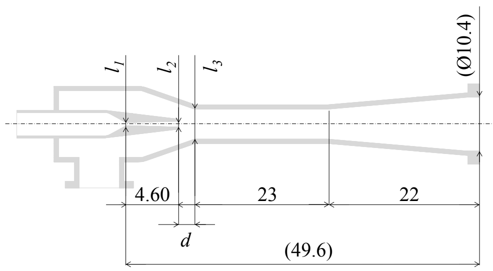

Fig. 4 Configuration of ejectors A and B

Table 1 Configuration parameters of ejectors A and B

	Nozzle throat diameter l_1	Nozzle exit diameter l_2	Mixing Section diameter l_3
Ejector A	1.40 mm	2.20 mm (A_{NE} = 2.45)	2.36 mm (A_{MIX} = 2.84)
Ejector B	1.40 mm	2.20 mm (A_{NE} = 2.45)	3.00 mm (A_{MIX} = 4.55)

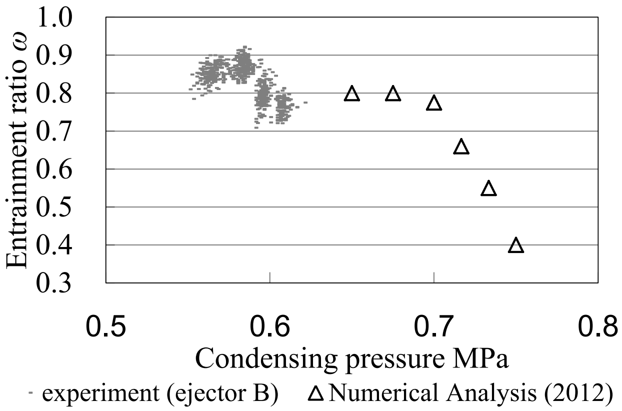

Fig. 5 Comparison of experimental and numerical analysis results for ω vs. condensing pressure P_c for ejector B at T_e = 15 °C, T_g = 60 °C

4.4. Effect of Nozzle exit

Table 2 shows the ejector configurations of new ejectors C and D. Those ejectors have a smaller size of the nozzle exit diameter than that of ejectors A and B. We calculated numerical analysis for different nozzle exit areas between 1.00 and 2.60 at a condition of T_e = 15 °C and T_g = 60 °C and a condensing pressure P_c = 0.718 MPa corresponding to about 27 °C with a mixing section area ratio of A_{MIX} = 4.55. From the calculation, A_{NE} = 1.26 can make HFC-134a to expand in a good manner without any shockwave [4]. Furthermore, the nozzle exit area ratio A_{NE} = 1.26 has a peak of entrainment ratio at P_c = 0.868 MPa corresponding to about 34 °C as the calculation result.

Figure 6 shows the experimental results of ω vs P_c in the case of ejector C at T_e = 15 °C and T_g = 60 °C. As shown in Figure 6, the experimental result does not agree with the result of the numerical analysis result. Experimental ω was smaller by about 0.25 from the calculation result and the critical pressure was lower about 0.2 MPa.

4.5. Comparison of four different types of ejectors

Figure 7 shows the experimental results of a relation between COP_T and T_c for ejectors A, B, C, and D at T_e = 15 °C and T_g = 60 °C. By enlarging the mixing section area, the COP_T of ejector A, whose configuration was designed on the basis of shock-circle model, improves as that of ejector B about double from 0.4 to 0.85. On the other hand, the T_c^* changes about 28 °C to 21 °C. It might be easy to understand that the efficiency improves by enlarging the mixing section area, while the critical condensing pressure is not automatically controlled at the best condition.

The highest T_c^* reaches to 29 °C in case of ejector D, whose nozzle exit diameter is 1.57 mm and the mixing-section diameter is 2.36 mm. A difference between ejectors C and D is a diameter of mixing section. By changing the diameter of mixing section, the performance changes from about 0.7 to 0.3 in COP_T and about 23.5 °C to 29 °C in T_c^*.

As a summary about the ejector configurations and the performances, it can be said that the COP_T and T_c^* cannot exceed the linear relation, which is drawn by a dashed line in Figure 7.

Table 2 The configuration parameter of ejector C and D

	Nozzle throat diameter l_1	Nozzle exit diameter l_2	Mixing Section diameter l_3
Ejector C	1.40 mm	1.57 mm (A_{NE} = 1.26)	3.00 mm (A_{MIX} = 4.55)
Ejector D	1.40 mm	1.57 mm (A_{NE} = 1.26)	2.36 mm (A_{MIX} = 2.84)

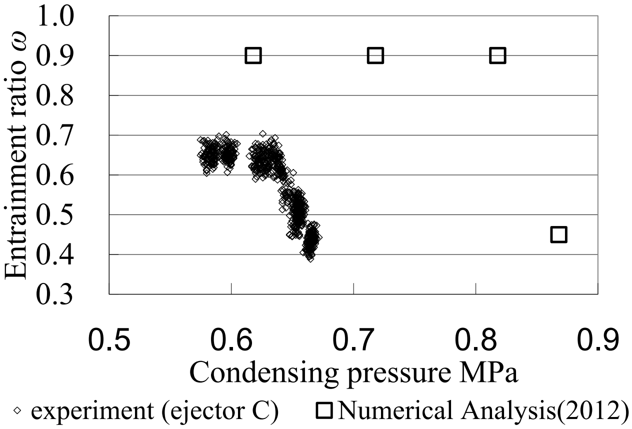

Fig. 6 Comparison of experimental and numerical analysis results for ω vs. condensing pressure P_c for ejector C at T_e = 15 °C, T_g = 60 °C

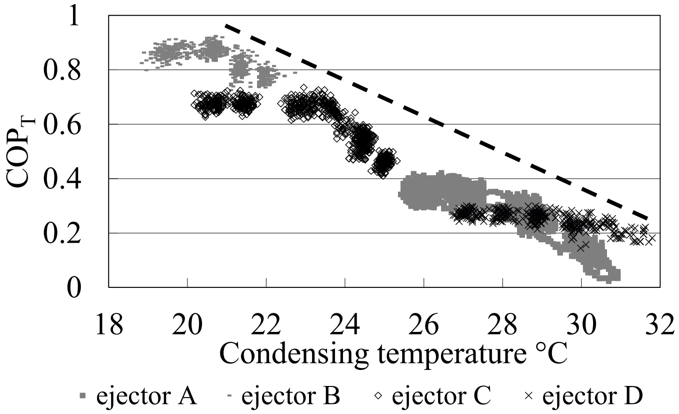

Fig. 7 Experimental results of COP_T for ejectors A to D at T_e = 15 °C, T_g = 60 °C

5. Conclusion

Ejector refrigeration cycle is a simple system, which can provide cooling by using solar thermal energy being higher than around 60 °C. The ejector configuration design is a key to get a high efficiency of the cycle. In order to analyze the behavior of the refrigerant in ejector, CFD and experimental test was applied to get the best geometry parameters through a trial and error approach. Four sets of configuration parameters of a mixing section area and a nozzle exit area of the ejector were tested. As a conclusion, an appropriate mixing section area can make the greatest cooling capacity but the condensing temperature changes, while a smaller nozzle exit area can avoid the energy loss of shockwave that makes reliable repeatability and higher condensing temperature but it is possible to be confirmed at conditions of lower cooling capacity in our experimental trials. The performances of ejector cycle having four different sets of configuration parameters cannot exceed a linear relationship between cooling capacity and condenser temperature so far. In the next step, we might be requested to search an appropriate combination of nozzle exit area, mixing-section area and the length of those parts, as well as the configuration of a diffuser part.

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